HIGH SPEED AND TEMPERATURE COUNTER-ROTATING INTERSHAFT SEALS FOR TURBINE ENGINES

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The Stein Seal company is developing counter rotating seals under an SBIR with the Navy Air Warfare Center. The COTR for the Navy is Pauline Tarrantini, and the Principal Investigator at Stein is Alan McNickle.

Counter- Rotating (C-R) shafts in turbine engines offer significant weight reduction and increased efficiency.. In a CR twin spool engine, the high speed shaft is supported directly by the low speed shaft via the counter rotating inter-shaft bearing. The discharge from the high-pressure turbine into a counter rotating low-pressure turbine eliminates the need for stator vanes to redirect the steam velocity as in co-rotating engines. The absence of a stator results in a reduction in overall engine weight, and the elimination and improved efficiency. Counter rotation causes high interface seal velocities and rubbing seals wear excessively. High centrifugal loads are imposed on all components of the seal. The seals must accommodate motions of both shafts which translates to large misalignments and dynamic excursions. Also, the available space between shafts, into which a seal must be inserted is limited. Radial movement due to centrifugal and thermal effects has led to large clearance requirements with conventional solid bushing configurations. As a consequence of large clearances, contemporary seals have high leakage and large heat exchangers and air oil separators are required. The Stein Seal concepts that were evaluated are intended to circumvent large clearance requirements and minimize leakage.

The operating conditions were 95 psia ,high pressure, 55 psia low pressure. The environmental temperature was 850 °F. The high speed shaft was running at 18,600 rpm, and the low speed shaft ran at 11,200 rpm. The target leakage was 0.5 SCFM/psid.

A number of seal configurations were considered. The hydrodynamic face seal consists of a full ring and a gap piston ring secondary seal that is centrifugally and spring loaded against the outer shaft causing the face seal to rotate with the outer shaft. The seal is pressure balanced by through holes from the high pressure to a balance groove on the low-pressure side. The leakage path is from the balance groove to the OD of the seal ring. The interface contains a hydrodynamic Rayleigh-step geometry. High pressure flows into a shallow step of about 10 mils deep and then flows into a deeper groove that is 30 mils deep that surrounds the shallow step with high pressure and thus improve hydrodynamic performance. Hydraulic loading of 700 lbs. is the principal closing mechanism. The OD is close to 6 inches and the ID is 5.04 inches. The operating clearance is 0.4 mils.

The free floating ring seal is split like a piston ring and is radially loaded against the outer shaft and rotates with it. The hydrodynamic interface is similar to the face seal. High pressure from the Rayleigh-step is connected to the OD to help balance the centrifugal and pressure radial loads.

The Centrifugal Compensating Seal was an invention of Dr. Philip Stein and was designed to compensate for centrifugal loading. The single piece balance ring mates with sectored components on either side. Centrifugal loading on the left components force the balance ring to the right to push the right components down. The entire design floats and does not rotate with either the OD or ID runner. It is designed to float at a speed that provides equal radial and axial clearance. The seal speed is half of the difference in operating speeds of the two shafts.

The final design was a balanced hydrostatic seal with a gap piston ring as the secondary seal. High pressure is ported into the interface that consists of 30 holes of 30 mil diameter. Spring loading maintains closure in the absence of pressure activation.

All the seals were face type seals because when the mating components rotate in opposite directions the hydrodynamic speed is the sum of the two speeds. For dynamic considerations, there is a major difference between a counter rotating seal and a normal face seal. Normally in a face seal there is a rotating rotor that mates with a passive non-rotating seal ring that tries to follow the excursions of the rotor. In counter rotation the seal ring housing also rotates and is exposed to additional vibratory excursions. The inner and outer rotors both have excursions in the X, Y and Z directions at different operating speeds, so that they can never be synchronized. They are sometimes in phase and sometimes out of phase. The dynamic analysis allowed relative axial motion between the seal ring and the outer shaft but not relative angular motion. This constraint is true for the ring seals, but the hydrodynamic face seal and the hydrostatic seal may have relative angular motion through the piston ring. Thus, the analysis is very conservative for the hydrostatic and hydrodynamic face seals. The results of the dynamic analysis were discouraging in that all seals except the hydrostatic seal could not follow rotor excursions without contact. The higher operating clearances of the hydrostatic seal allowed rotor tracking at reasonable levels of rotor misalignment. The design clearance of the hydrostatic seal was 1.5 mils. The leakage was approximately 0.25 mils/psid or approximately half of target values. With 1 mil misalignment of both rotors, the minimum film thickness was slightly below 1 mil, which is quite acceptable.

A phase 2 SBIR has been received to develop and test a hydrostatic seal in a counter rotating rig.

Questions

Q. What will the centripetal field do to the closing springs?
We haven't calculated that, but I don't expect it will be very much. We will look at it in Phase II.

Q. What is the rotary speed of the seal? Is it the sum or difference of the two shafts.
A. The sum of the two shafts, which is about 30,000 rpm. The springs will rotate at the outer shaft speed.

Q. What is the power consumption?
A. At the 1.5 mil condition, not quite 0.5 hp.

Q. Did you use constant coefficients for the dynamic analysis?
A. Yes. Thus, the analysis is conservative in that the stiffness does not increase as a function of closure.

Sponsoring Agency
Naval Air Warfare Center
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- COTR - Pauline Tarrantini
- Principal Investigtor - Alan McNickle

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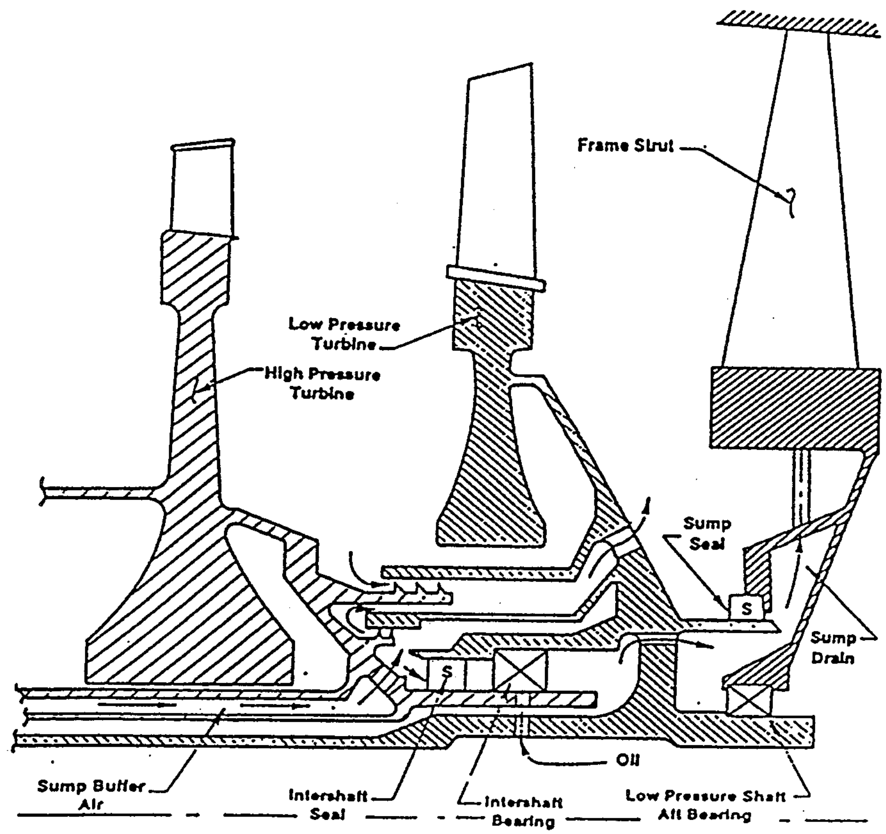
Intershaft Seal Location



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Advantages of Counter Rotation

- Less Weight
  - Stator not required between HP and LP turbines
  - Low leakage seals will reduce weight of heat exchangers and oil separators
- Improved Efficiency
  - Elimination of Stator Vanes

Difficulties of Counter Rotation

- High inteface speeds for bearings and seals
- Centrifugal loading
- Dynamic excursions of two shafts

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Operating Conditions

Upstream Pressure	95 psia
Downstream Pressure	55 psia
Temperature	850 Deg. F
HP Turbine Speed (Inner)	18,600 rpm
LP Turbine Speed (Outer)	11,200 rpm
Gas Viscosity	4.96E-09 lb-s/in**2
Gas Constant	247,605 in**2/s**2/deg F
Target Leakage	0.5 SCFM per psid

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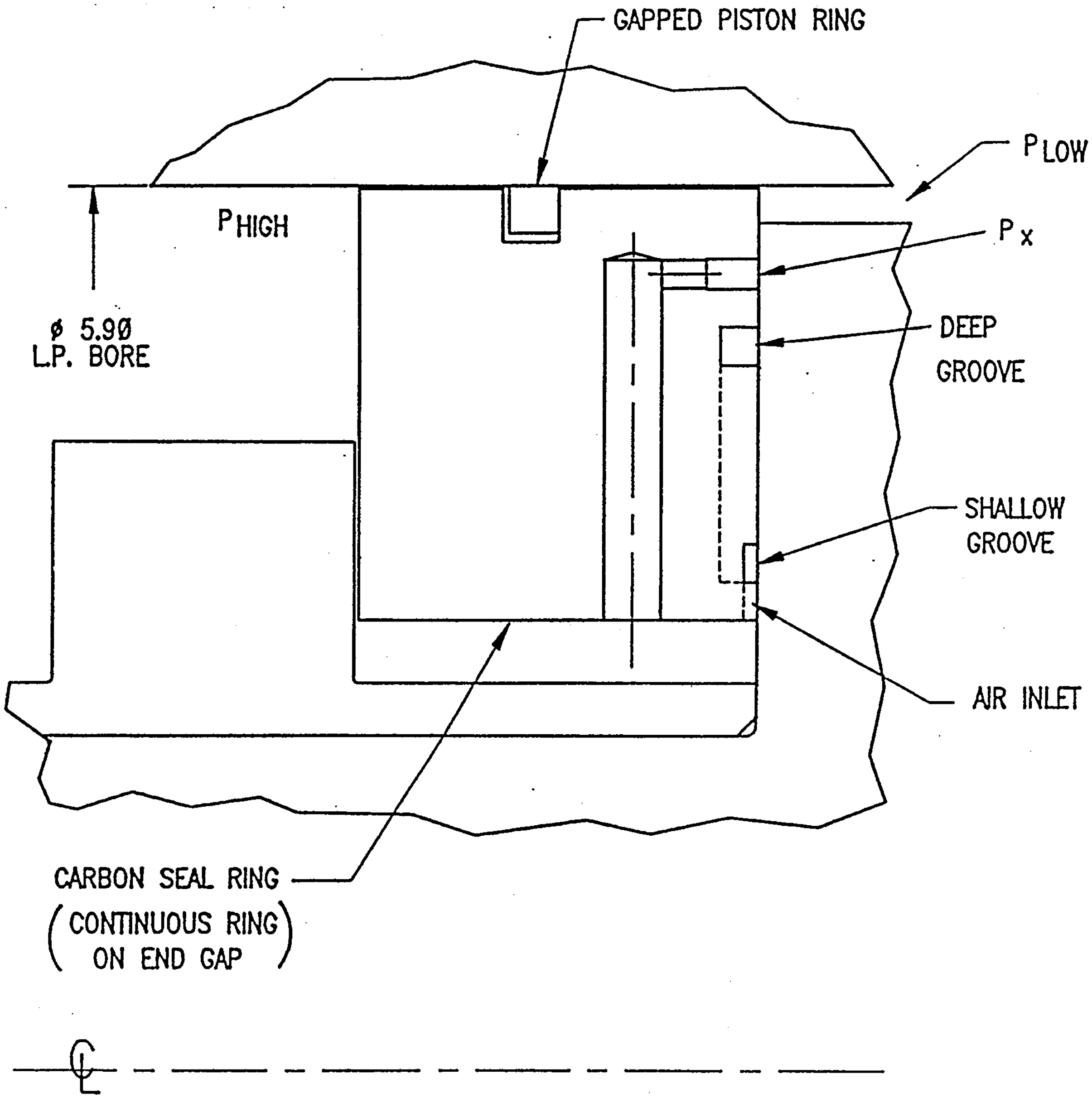
Designs Considered

- Hydrodynamic Face Seal
- Free Floating Ring Seal
  - Single and Multiple Sectors
- Centrifugal Compensated Seal
- Balanced Hydrostatic Face Seal

Hydrodynamic Face Seal



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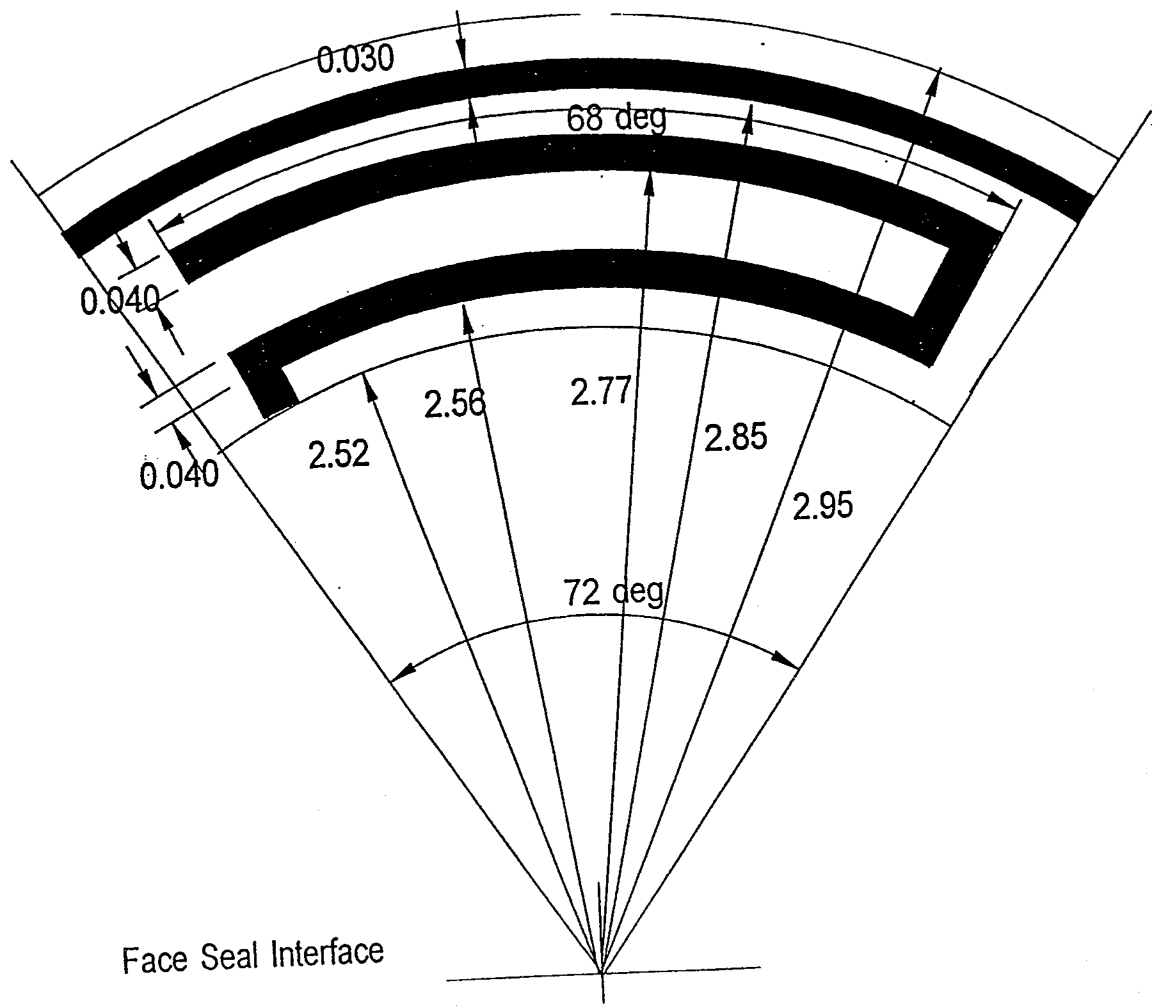
Face Seal Hydrodynamic Geometry



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Face Seal Parameters

OD	5.9 in.
ID	5.04 in.
Clearance	0.4 mils
Load	702 lbs
Viscosity	4.957 e-09 reyns
Temp.	850 F
High Pressure	95 psia
Low Pressure	40 psia
Shallow Step Ht.	0.010 in.
Deep Step Ht.	0.030 in.

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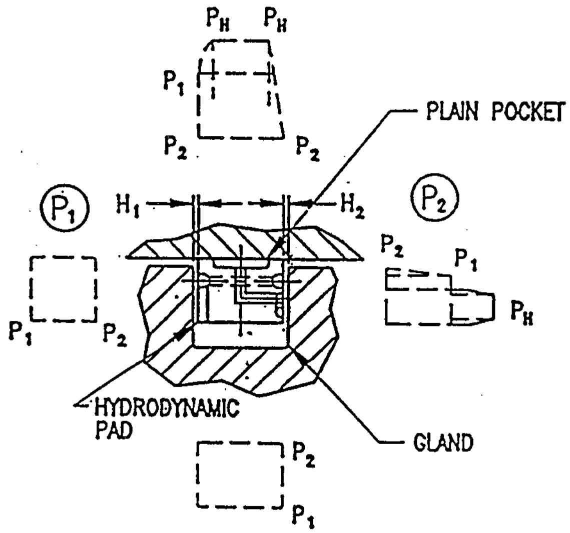
Free Floating Ring Seal



NOTES:

1). PRESSURE PROFILES SHOWN FOR $H_1 > \emptyset$ & $H_2 = \emptyset$

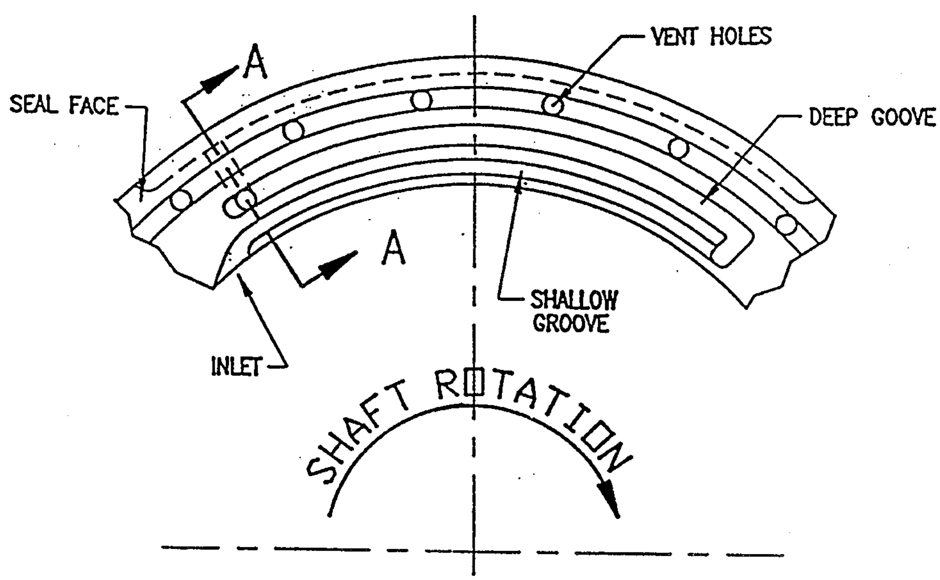
2) P_1 = HIGH PRESSURE

3) P_2 = LOW PRESSURE

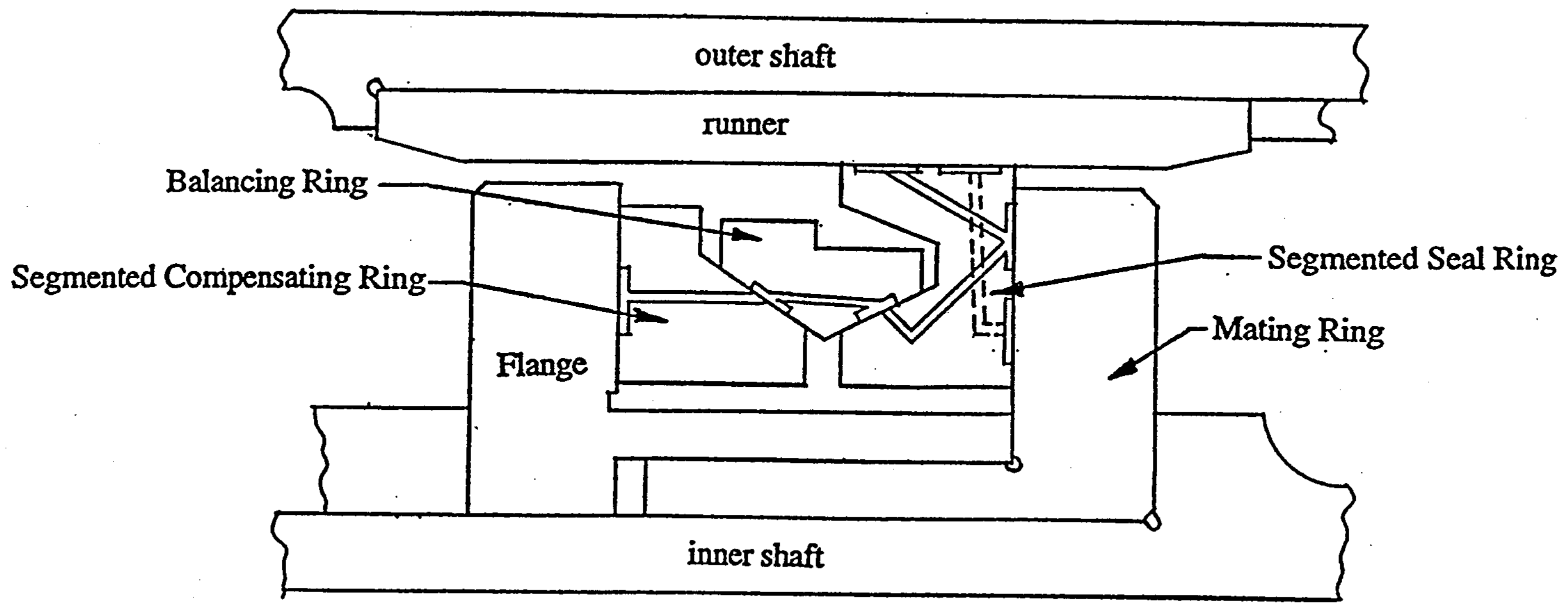
4) P_H = HYDRO DYNAMIC PRESSURE

5) SHADED AREAS ARE RECESSED.

SECTION A—A



Centrifugal Compensating Seal



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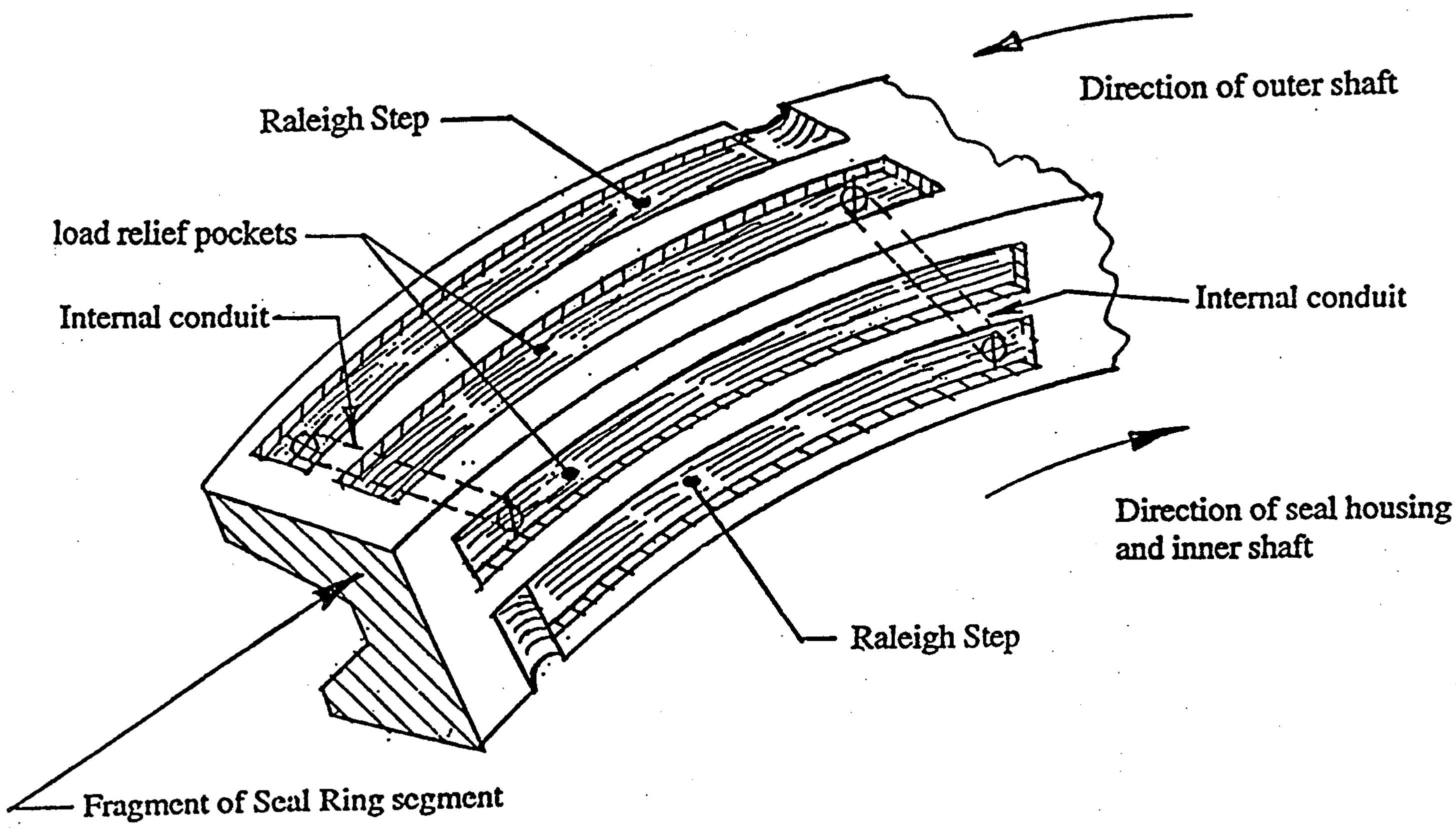
Seal Ring Bearing Arangement



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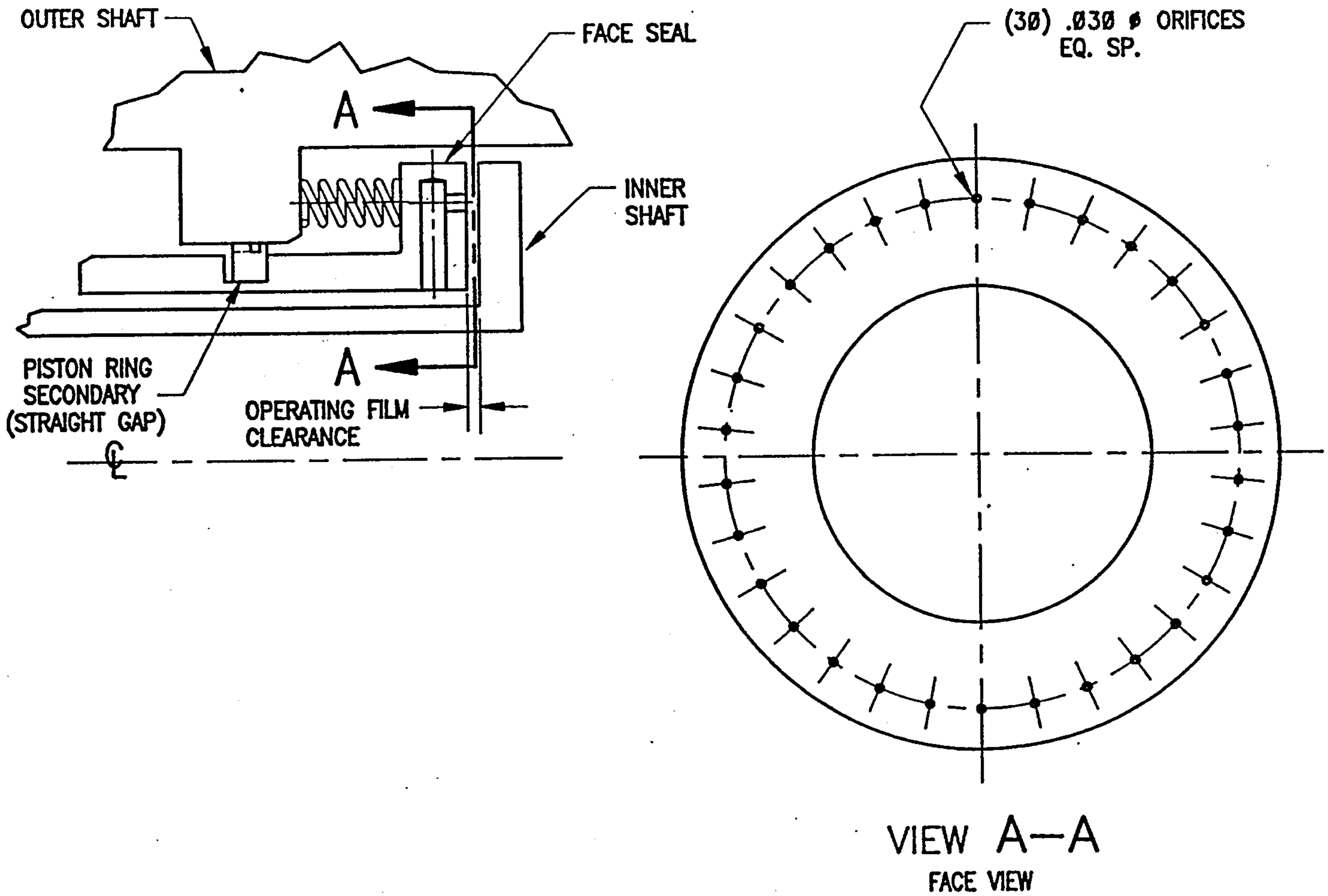
Balanced Hydrostatic Face Seal



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Hydrostatic Seal Parameters

OD	5.84 in
ID	5.04 in.
Viscosity	4.96e-09 Reyns
Temperature	850 F
Speed	29,800 rpm
High Pressure	95 psia
Low Pressure	40 psia
Orifice	0.02, 0.03, 0.04 in.

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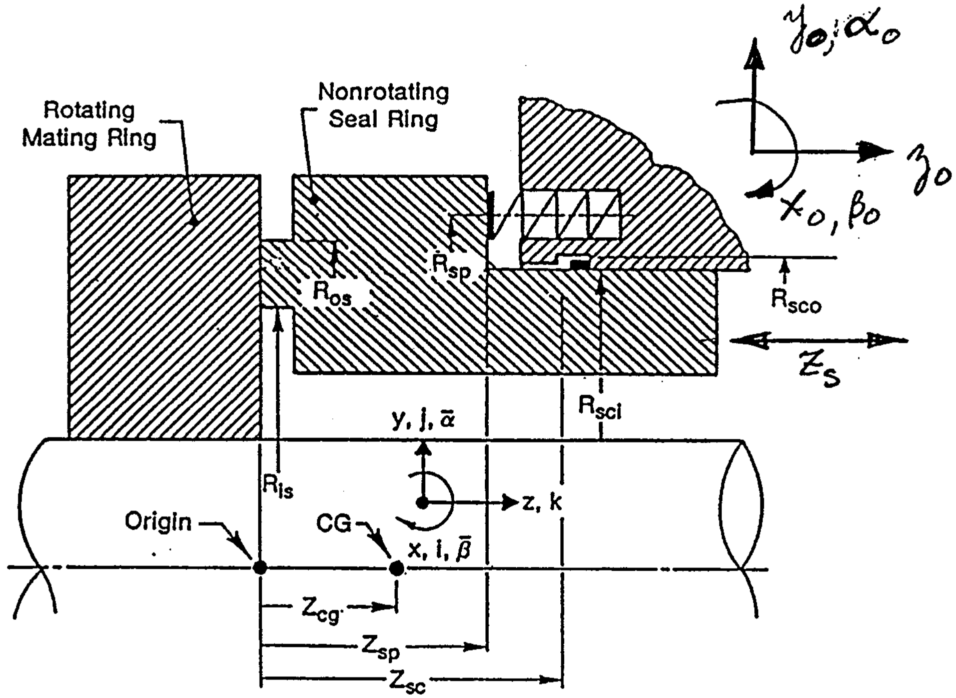
C-R Dynamic Model



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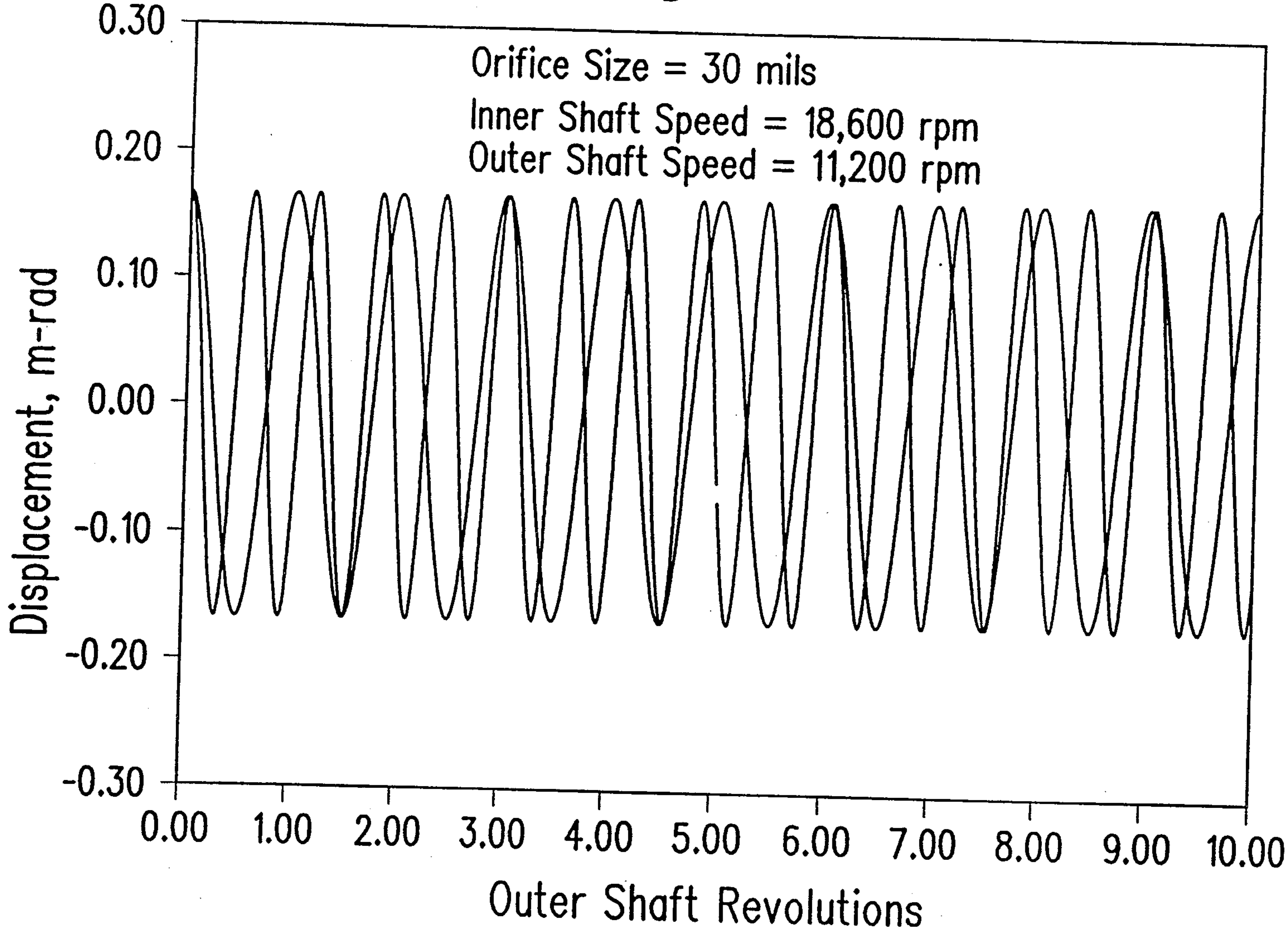
Shaft Dynamics



Shaft Angular Modes

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Face Seal Dynamics - 1 DOF DYCOUNT

f	Zo, Zi, mils	TIR Each Shaft, mils	Result
0.2	0.25	0.001	F
0.2	0	0.001	F
0.2	0	0.0005	F
0	0	0.0005	F
0	0	0.00025	T

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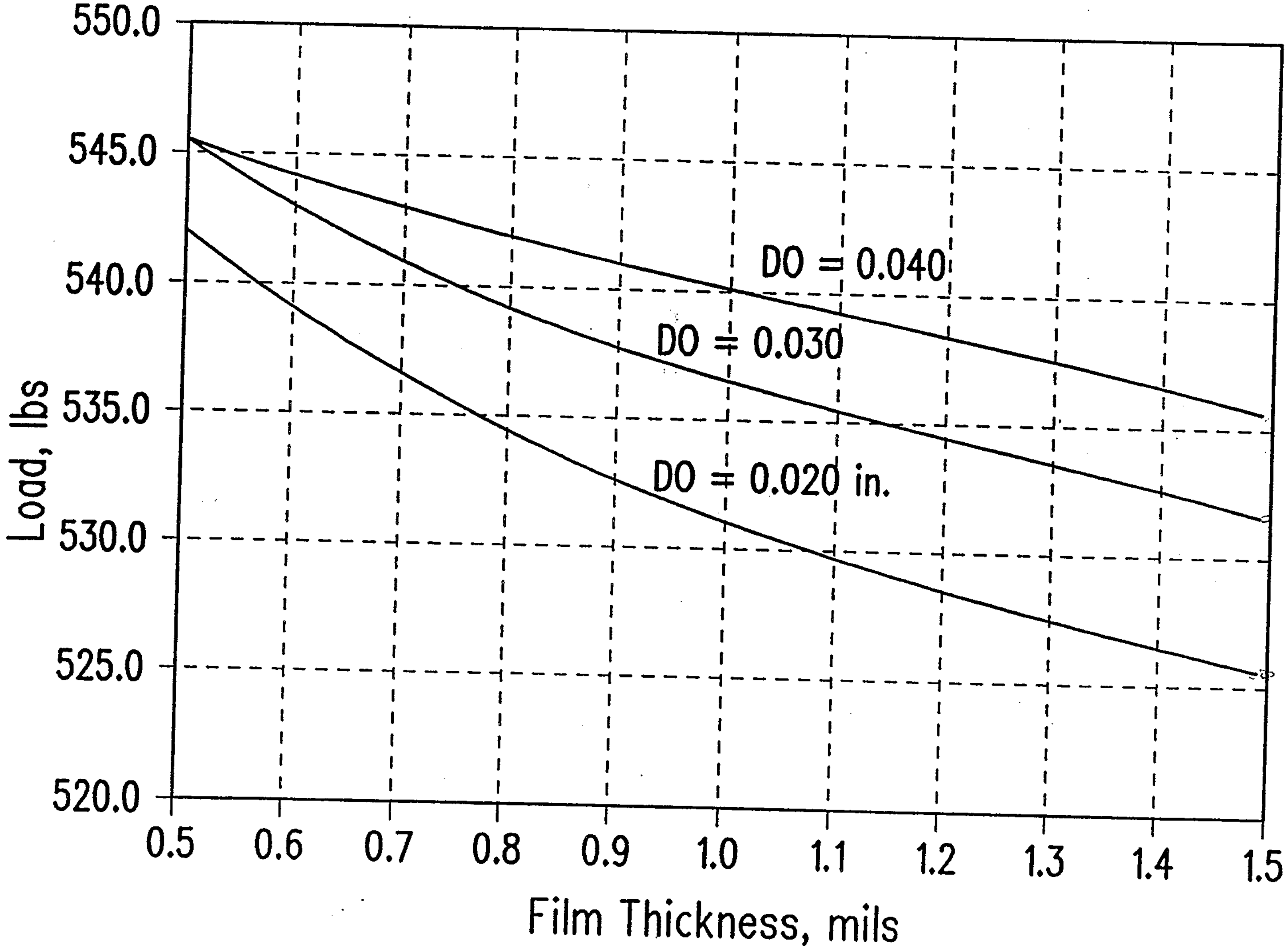
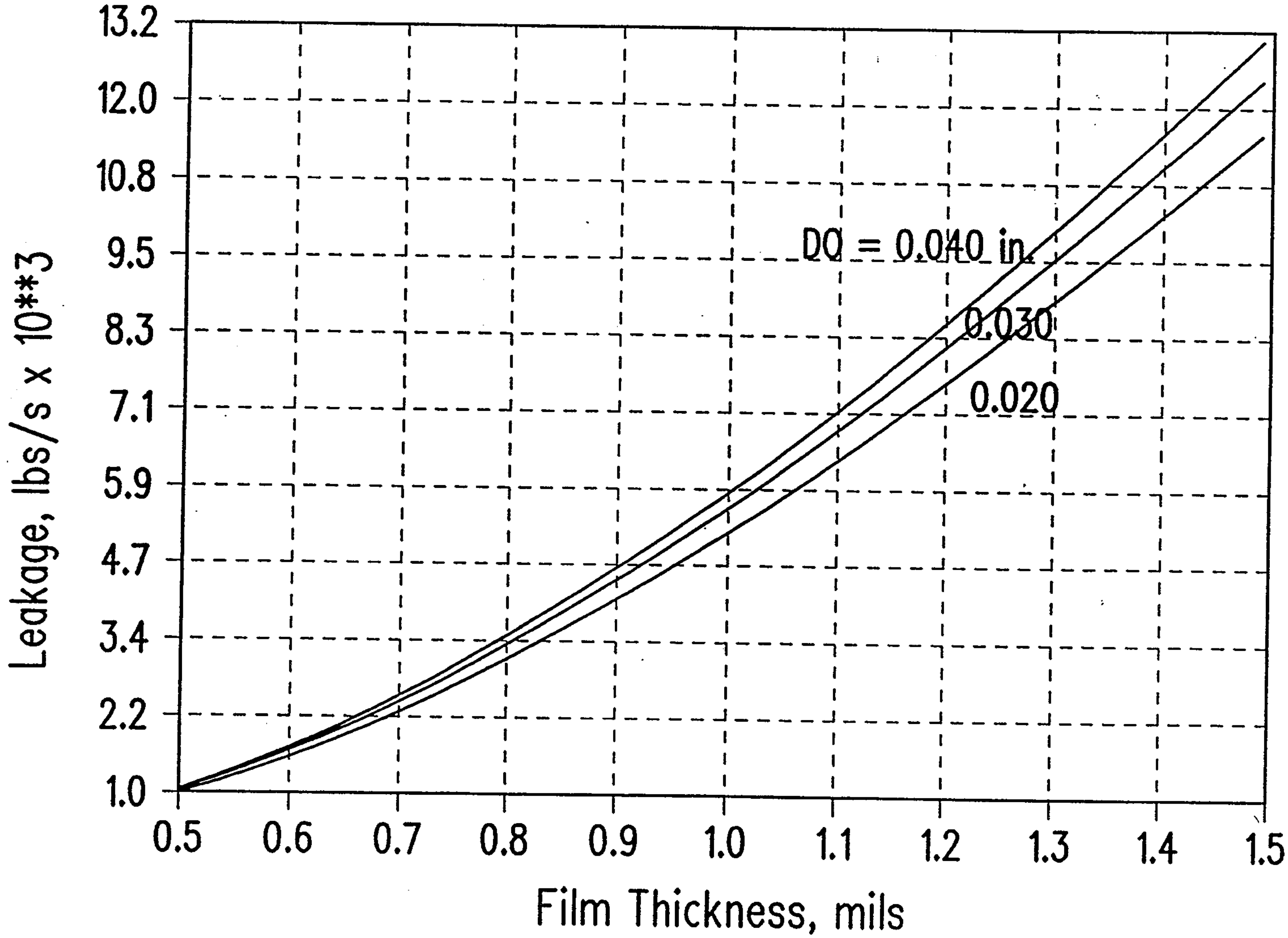
Hydrostatic Seal Load Capacity
Clearance = 1.5 mils



Hydrostatic Seal - Leakage



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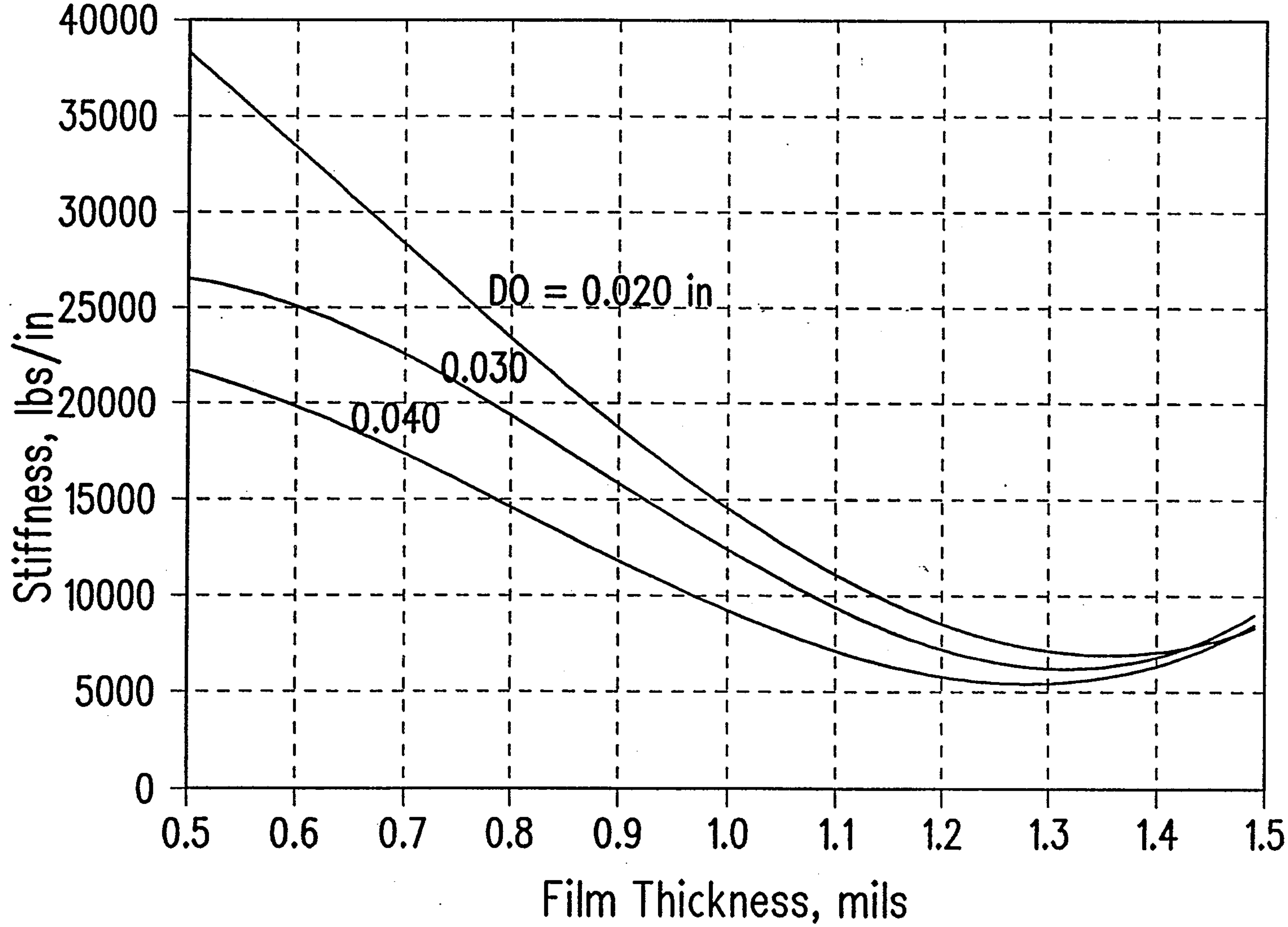
Hydrostatic Seal - Leakage

Orifice Size, in.	Leakage, SCFM	Leakage Ratio SCFM/psid
0.020	9.212	0.23
0.03	9.898	0.247
0.04	10.401	0.260

Hydrostatic Seal Stiffness



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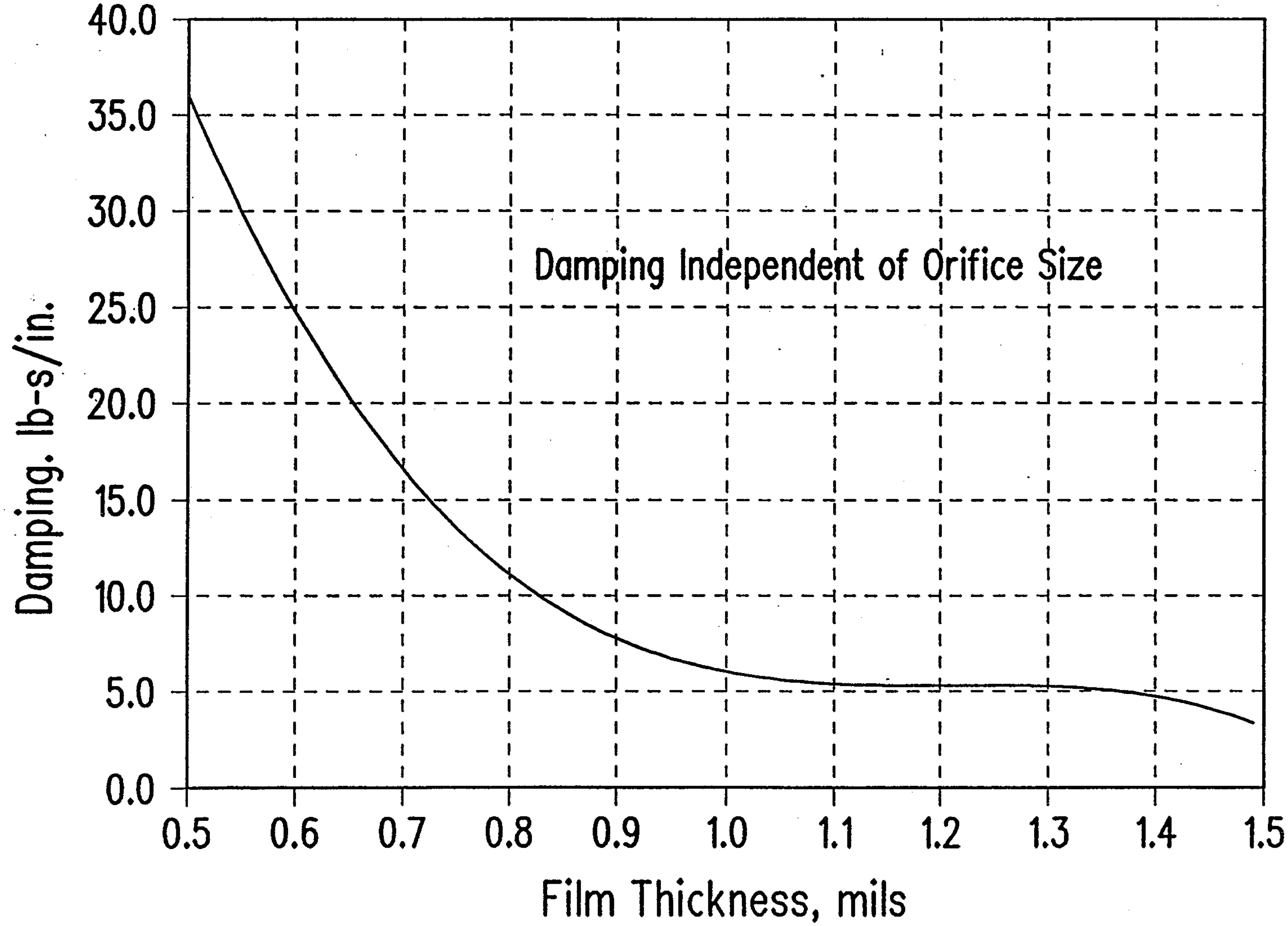
Hydrostatic Seal - Damping



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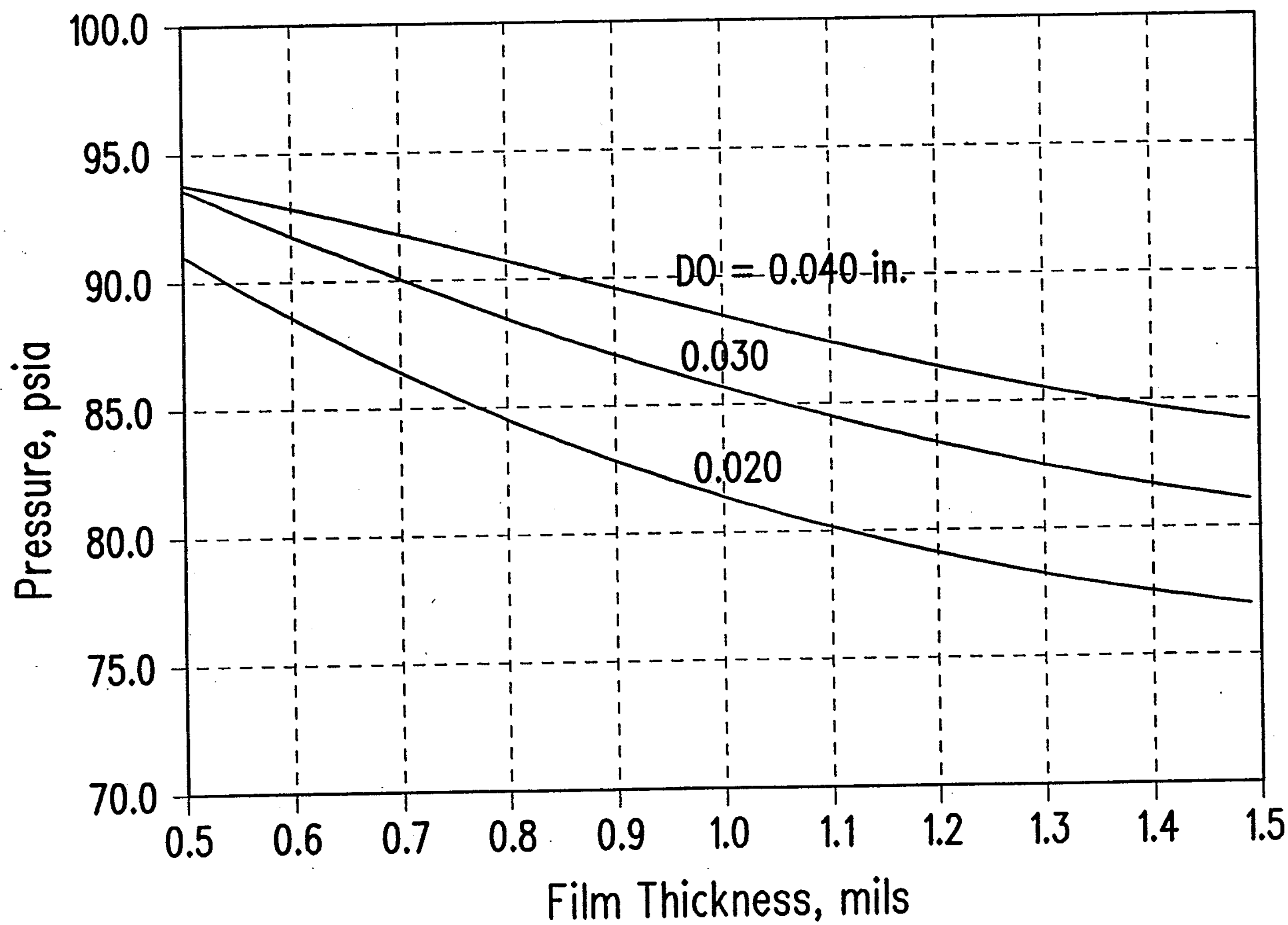
Hydrostatic Seal - Downstream Pressure



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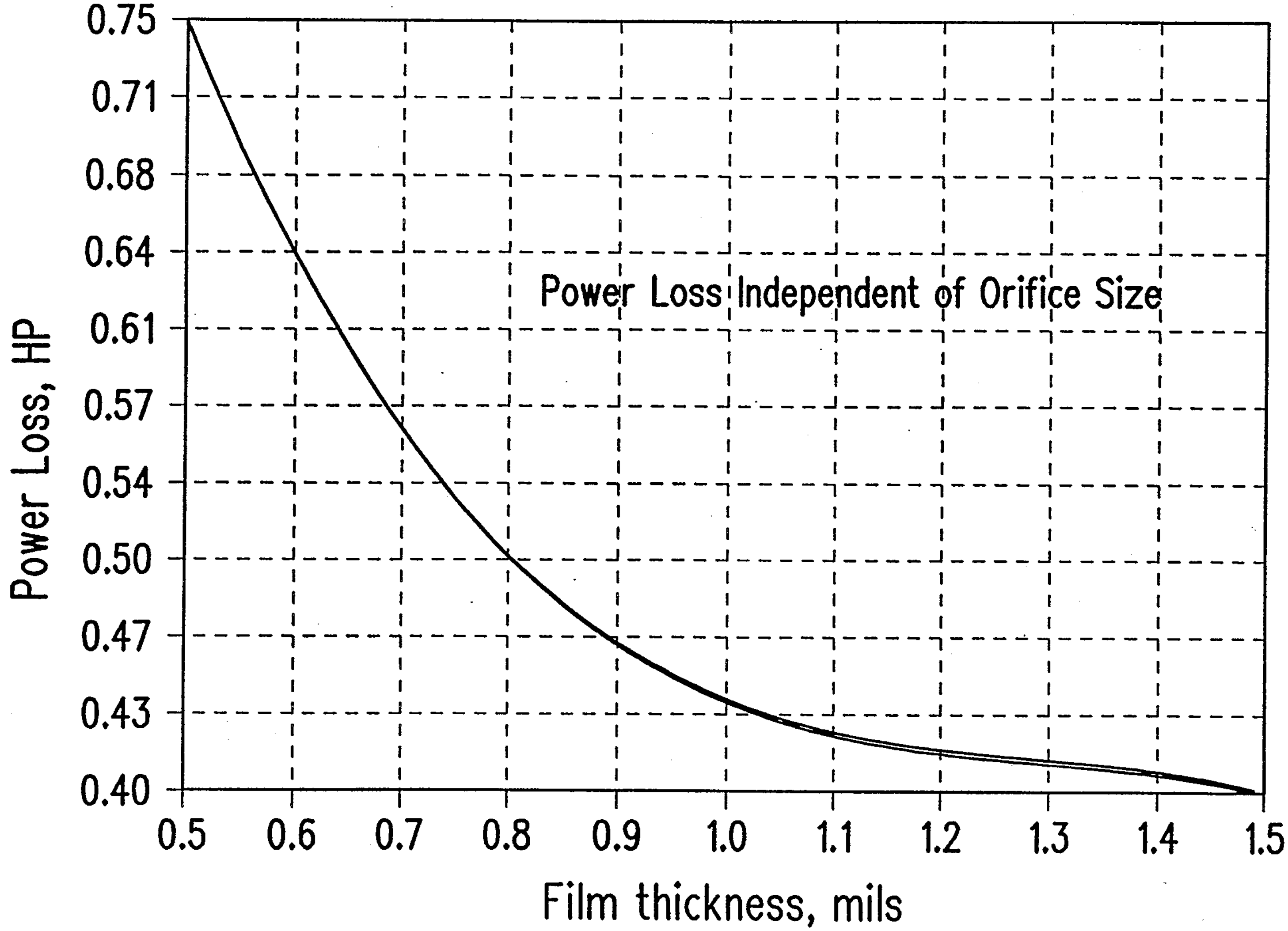
Hydrostatic Seal - Power Loss



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Hydrostatic Seal - Stiffness Clearance = 1.5 mils

Kzz	9995 lb/in
Kza	21.17 lb/rad
Kzb	21.17 lb/rad
Kaz	0 in-lb/rad
Kaa	58,830 in-lb/rad
Kab	4298 in-lb/rad
Kbz	0 in-lb/in
Kba	-4298 in-lb/rad
Kbb	58,830 in-lb/rad

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Hydrostatic Seal - Damping Coefficients

Dzz	3.32 lb-s/in
Dza	0 lb-s/rad
Dzb	0 lb-s/rad
Daz	0 in-lb-s/in
Daa	11.79 in-lb-s/rad
Dab	-.22 -n-lb-s/rad
Dbz	0 in-lb-s/in
Dba	.22 -n-lb-s/rad
Dbb	11.79 in-lb-s/rad

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Hydrostatic Seal - Rotor Excursions

Outer Shaft Axial Amplitude	0.25 mils (0.5 mils TIR)
Inner Shaft Axial Amplitude	0.25 mils (0.5 mils TIR)
Outer Shaft Angular Rotation Amplitude	1.67 x 10-04 rad. (1 mil TIR)
Inner Shaft Angular Rotation Amplitude	1.67 x 10-04 rad. (1 mil TIR)

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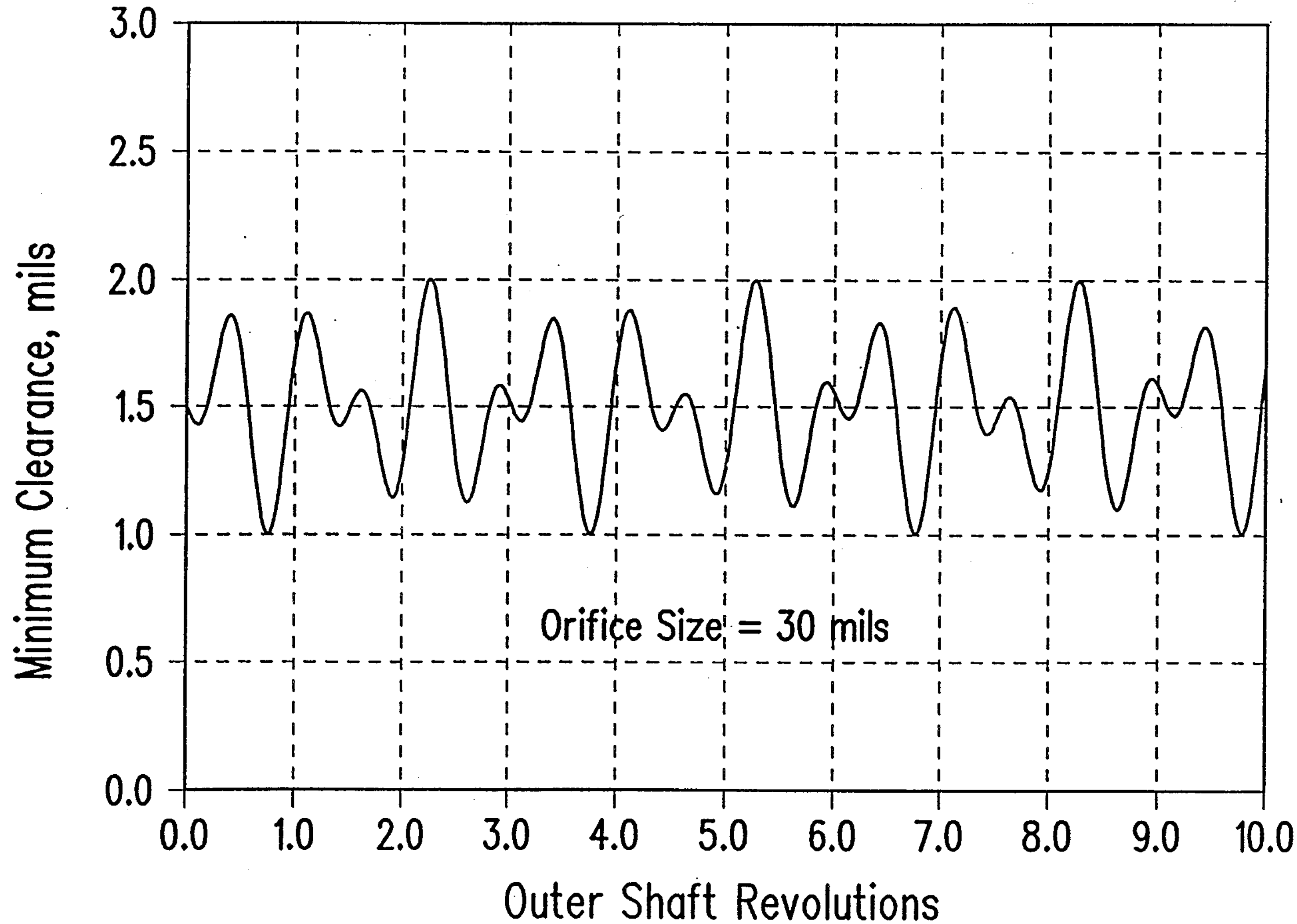
Hydrostatic Seal - Minimum Clerarance



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Summary-Results and Conclusions

- Principal problem is dynamic tracking
  - Two rotors at different speeds
- Balanced Hydrostatic Seal is favored candidate because of tracking capability
  - High clearance operation compared to hydrodynamic configurations which operate at low film thickness

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Further Research

- Determine Coefficient of Discharge for hydrostatic configuration.
- Test seals on C-R rig.

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